

Experimental Validation of Boundary Element Methods for Noise Prediction

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ABSTRACT

Experimental validation of methods to predict radiated noise is presented in this paper. A combined finite element and boundary element model was used to predict the vibration and noise of a rectangular box excited by a mechanical shaker. The predicted noise was compared to sound power measured by the acoustic intensity method. Inaccuracies in the finite element model shifted the resonance frequencies by about 5 percent. The predicted and measured sound power levels agree within about 2.5 dB. In a second experiment, measured vibration data was used with a boundary element model to predict noise radiation from the top of an operating gearbox. The predicted and measured sound power for the gearbox agree within about 3 dB.

INTRODUCTION

The prediction of noise in the design stage is important for building low-noise machinery. In the past this was seldom done because the analysis techniques lacked the necessary accuracy and detail. Except for the use of simple rules of thumb, the designer's best guide was often years of experience and a large database of old noise data. To shorten the time to bring a product to market, companies are now beginning to rely less on testing and more on analysis. However, the designer must have confidence in the analysis tools used. Confidence may be achieved in a variety of ways including comparison of predicted data with test results. In this paper the acoustic boundary element (BE) method is validated by comparing the predicted sound power of a test gearbox with that measured using the sound intensity method. The vibration of the gearbox was measured and used as input to the BE model.

An additional step in the prediction of noise occurs if the vibration is predicted rather than measured using, for example, a finite element (FE) model of the structure. In this paper a simple box-like structure was used to test the feasibility of using a combined FE/BE model to predict both structural vibration and radiated sound power.

PREDICTION OF NOISE OF TEST GEARBOX

Figure 1 shows the NASA Lewis gear noise rig. The rig consists of a single-mesh gearbox driven by a 150 Kw variable speed electric motor and loaded by an eddy-current dynamometer [1]. Figure 1b shows the measurement of vibration and sound intensity on the top surface of the gearbox. Also shown in this figure is the sound measuring apparatus. Only the top surface of the gearbox was used to simplify and shorten the validation study. A pair of accelerometers was used to measure vibration; one accelerometer was moved from point to point and the second accelerometer was fixed to maintain a constant phase datum.

The acceleration data were acquired by a two-channel spectrum analyzer, transferred to a desktop computer, and converted to real and imaginary components of velocity. These velocity components were used to produce an input file for the BEMAP program [2]. For this work, a public-domain PC version of BEMAP was used.

The vibration measurements were made on a rectangular grid of 7x9 points, as shown in Fig. 1b. This grid was used in BEMAP to construct a boundary element mesh of 48 linear elements; each element consisting of four nodes, one at each corner. Although BEMAP also contains the more efficient quadratic elements, the linear quadrilateral element was selected because it matched the measurement grid on the top surface of the test gearbox. The rule of thumb for linear elements is that the mesh size should not exceed one-quarter of a wavelength for the highest frequency of interest. The nodal spacing was 38 mm; hence, the highest frequency meeting this requirement strictly is 2250 Hz.

Figure 2 shows the comparison of the sound power measured using the sound intensity system in Fig 1b [3] and that predicted from measured vibration using BEMAP. The data in Fig. 2 were obtained for a running condition of 3000 rpm and 68 Nm torque. Agreement is generally good throughout the frequency range, even above 2250 Hz, the limit suggested when using linear boundary elements.

A COMBINED FE/BE MODEL FOR NOISE PREDICTION

A box-like apparatus (Fig. 3) resembling the gearbox in Fig. 1 was constructed to evaluate the feasibility of using a combined FE/BE model for noise prediction. The apparatus consisted of sides and bottom of 12.7 mm steel plate and a top surface of 1.6 mm aluminum plate [4].

Random excitation was applied to the apparatus in Fig. 3. The sound intensity was measured on a hemispherical surface over the apparatus which was placed on the floor of the laboratory. In addition, the force applied to the top surface of the apparatus by the shaker was measured, and the measured sound power was normalized by this force for comparison to the predicted sound power based on a unit force.

An FE model was constructed for only the top surface of the apparatus in Fig. 3. The sides and bottom were not modeled as they were assumed to be perfectly rigid. The vibration level of the top surface was calculated for a unit input force using the ANSYS program. A software interface was used to transfer the FEM geometry and vibration data from ANSYS to BEMAP for the prediction of sound power per unit force. A BE model (Fig. 4) for the apparatus was constructed using quadrilateral and triangular elements. The BE model included the sides of the apparatus, even though the FE model did not. Thus, the three-dimensional geometry of the apparatus was modeled, but any sound radiated from the sides was neglected in the FE/BE model (i.e., in the BE model the nodes on the sides of the apparatus had zero velocity). Measurements showed that the vibration amplitude of the sides of the box was approximately 10 dB below that of the top surface.

Figure 5 shows a comparison between the measured sound power and that predicted from the FE/BE model. Note the approximately 5 percent power shift in the resonance frequencies due to the overly stiff FE model. Clamped boundary conditions were assumed in the FE model of the top surface of the apparatus. In the actual apparatus the plate forming the top surface was restrained along its perimeter by 12.7 mm square steel rods.

Figure 6 shows the measured and predicted sound power data from Fig. 4 summed into one-third octave bands. Also shown in this figure is the sound power predicted from measured vibration. The measured vibration data were obtained only for the top of the apparatus in a manner similar to that described for the gearbox. The table below shows the overall linear and A-weighted sound power levels for the three spectra in Fig. 6.

	A-weighted	Linear
Measured	113.5	120.5
Predicted (measured vibration)	112.0	119.0
Predicted (FE vibration)	109.5	118.0

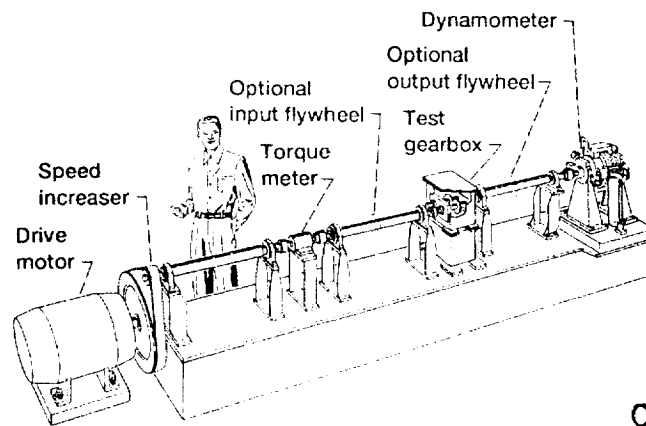
DISCUSSION OF RESULTS

It may be seen from the results that it is feasible to predict sound power quite reliably if the vibration data are known accurately. For example, when measured vibration data is used in BEMAP, the sound power may be predicted with an error of less than 2 dB overall. This is seen in the table above and in the results of Figs. 2 and 5. These predictions are quite good when one considers the host of experimental errors due to changes in operating conditions, drift in calibration, etc., over the several hours needed to measure the vibration data and the fact that the sound radiated by the sides of the apparatus in Fig. 3 was neglected in the prediction.

The results also demonstrate the need for accurate FE models for accurate prediction of machinery noise. Reliable estimates of damping on a per modal basis are important if the modes are underdamped as is the case for the apparatus in Fig. 3. (In practice, however, the damping is usually much higher for most built-up structures, thereby making the resonant response less critical). In addition, the structural boundary conditions must be known accurately. For the FE model of the apparatus in Fig. 3, it was assumed for simplicity that the sides of the box were rigid and the boundary conditions of the top plate were clamped. Clearly, these boundary conditions are only approximately fulfilled in the real apparatus.

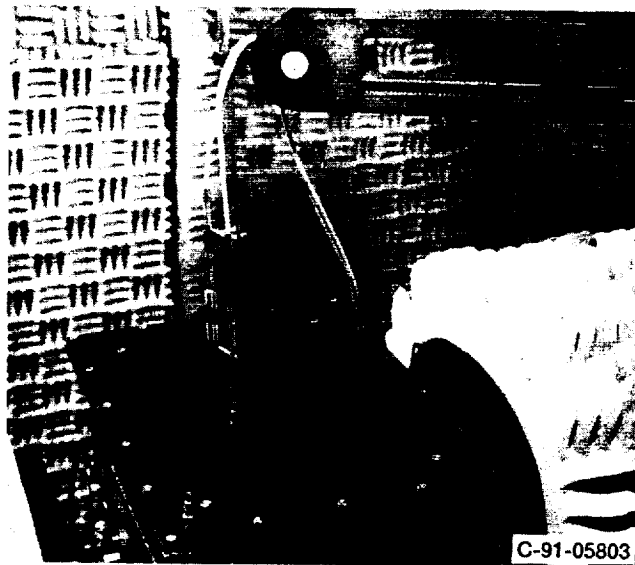
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1. "Comparison of Analysis and Experiment for Gearbox Noise," Fred B. Oswald, A. F. Seybert, T. W. Wu, and William Atherton, NASA TM-105330, 1992.
2. "BEMAP User's Manual," A. F. Seybert, T. W. Wu, and G. K. Wan, Spectronics, Inc., 1991.
3. "Automated Acoustic Intensity Measurements and the Effect of Gear Tooth Profile on Noise," W. J. Atherton, A. Pintz and D. G. Lewicki, NASA TM-100155, 1987.
4. "Validation of FEM and BEM to Predict Noise in Gearbox Structures," X. F. Wu, Ph.D. dissertation, University of Kentucky, 1992.



(a) Layout.

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(b) Test gearbox showing measurement grid and RAIMS robot with sound intensity probe.

Figure 1.—NASA gear-noise rig.

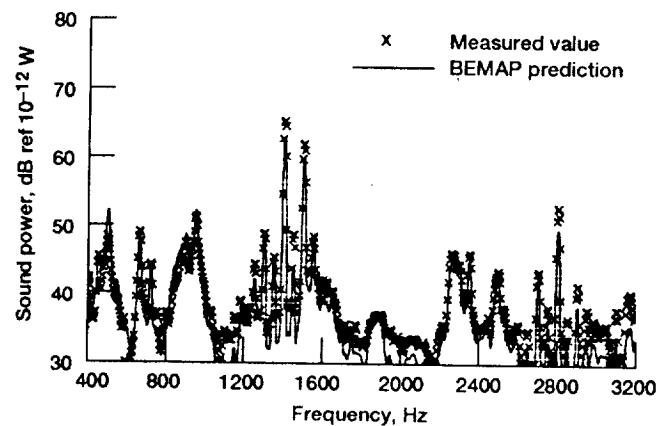


Figure 2.—Measured and predicted sound power levels for top of gearbox.

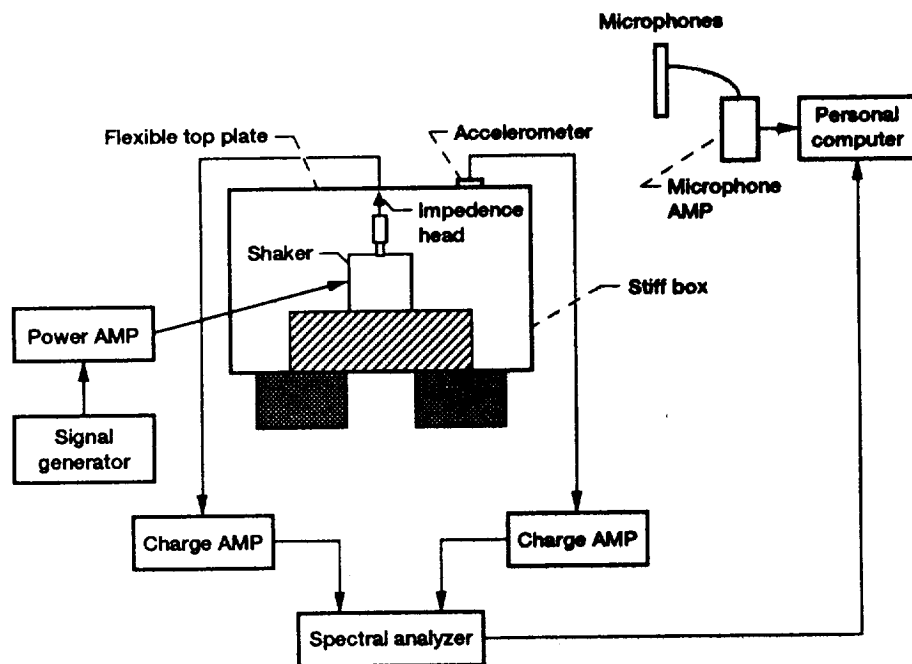


Figure 3.—Apparatus to verify FEM/BEM model.

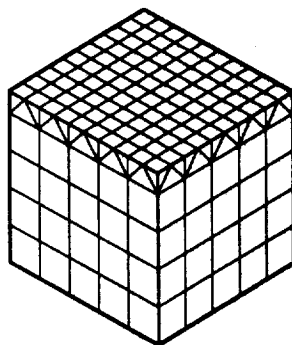


Figure 4.—Boundary element mesh for apparatus in Fig. 3.

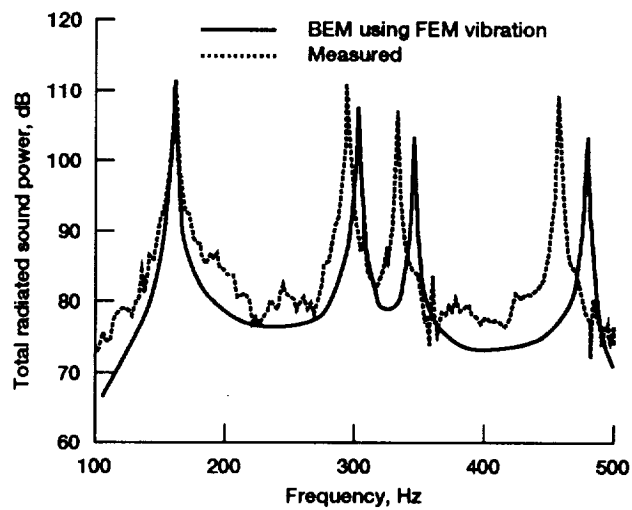


Figure 5.—Measured and predicted sound power level using FE/BE model.

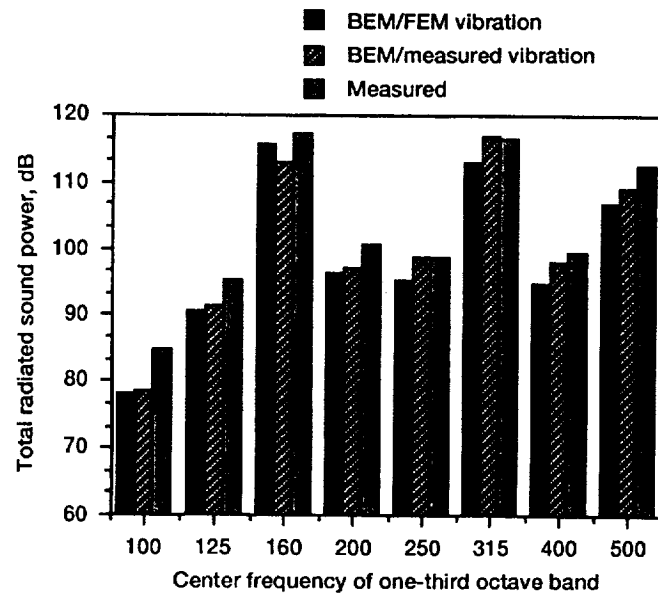


Figure 6—Measured and predicted sound power levels in one-third octave bands.

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